SOMETIMES IT GETS PUSHED

In the spring of 1954 Ed Eckert was handed a hot job. The Caterpillar DW 15 scraper type vehicle shown in Figure 1 had been on the market for about 6 months and already complaints about failed axle shafts had become too numerous. (Figure 2 shows a failed DW 15 axle shaft.) These complaints came from several customers, and in particular, from one customer in the area of Baltimore, Maryland.

Every day the scrapers were used with the present axles the risk of more failures became greater. It would be difficult to improve the strength of the axles without increasing their size. This would require larger bearings and hence larger axle housings, a costly and embarrassing remedy for the world's largest manufacturer of earth moving equipment.

Ed Eckert graduated in Agricultural Engineering from Iowa State University and started to work for Caterpillar as his first job in 1950. His first assignment was at the Caterpillar Proving Grounds as a test engineer. In 1954 he was transferred to the Applied Mechanics Division of the Research Department. As part of his assignment he worked on the stress analysis of rocker arms. During his first few years at Caterpillar he took some graduate courses from the University of Illinois. The courses had been arranged at a convenient location for Caterpillar employees.

The first thing Ed wanted to do when he was given the DW 15 job was to find out why the axle shafts were failing in service when they had held up fine in the tests to which Caterpillar subjects all of their new products.

 Written by Robert H. Todd with the cooperation of E. J. Eckert and the Caterpillar Tractor Company. Published with financial support from the National Science Foundation.

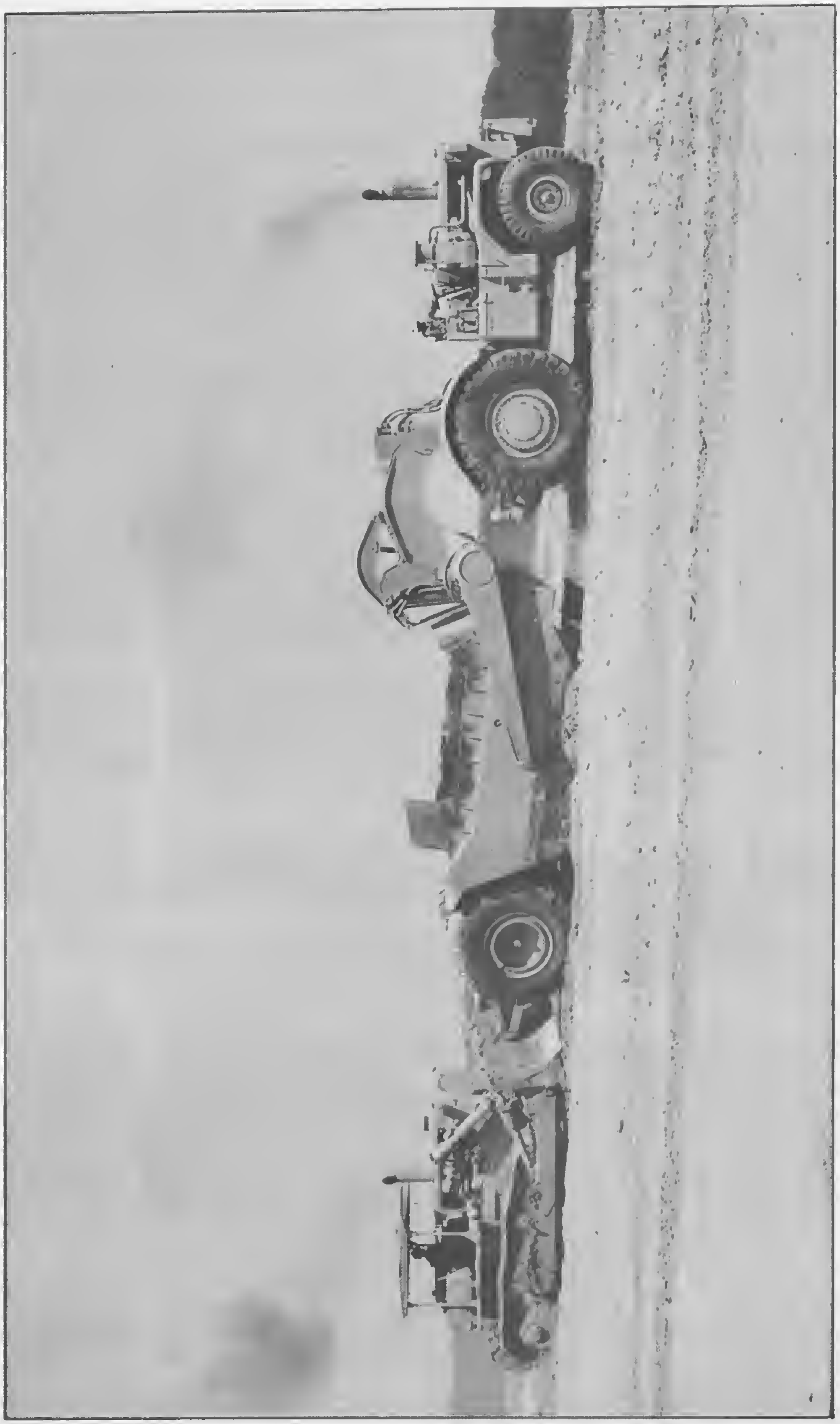


Figure 1

DW 15 Tractor With Bowl Scraper Being Assisted by a Push Tractor

Figure 2

A Failed DW 15 Axle Shaft

A drawing of the axle shaft is shown in Figure 3. Ed decided to take such a shaft off the shelf, apply some strain gages and use the instrumented shaft as a torque meter in a DW 15 tractor during regular field service. "I'm a great believer in trying to find out what the actual loads in service are", said Ed. He collected the necessary equipment for field testing in his car and started out for Baltimore together with Marty Kloet, another Caterpillar engineer, who worked in the Metallurgical Department. The field testing equipment consisted of the instrumented DW 15 axle shaft, (see Figure 4), a 110 Volt generator unit, a Brush recorder, an amplifier, and several mercury slip rings.

Once Ed and Marty arrived in Baltimore Ed mounted the instrumented axle shaft which they brought with them into one of the DW 15 tractors which was being used with a bowl scraper. The necessary recording equipment was bolted onto a piece of plywood and the plywood was attached to the fender of the tractor with C clamps. When Ed and Marty analyzed the data they could see what was happening. When the DW 15 needed some assistance from a push tractor, (see Figure 1), the torque in the axle shafts of the DW 15 was reversed. As the DW 15 was being pushed sometimes the wheels would slip. This reverse slipping of the DW 15's wheels caused its axles to experience a fully reversed torque. From the strain gage readings it was determined that this maximum reverse torque amounted to about 50,000 ft.-lbs.

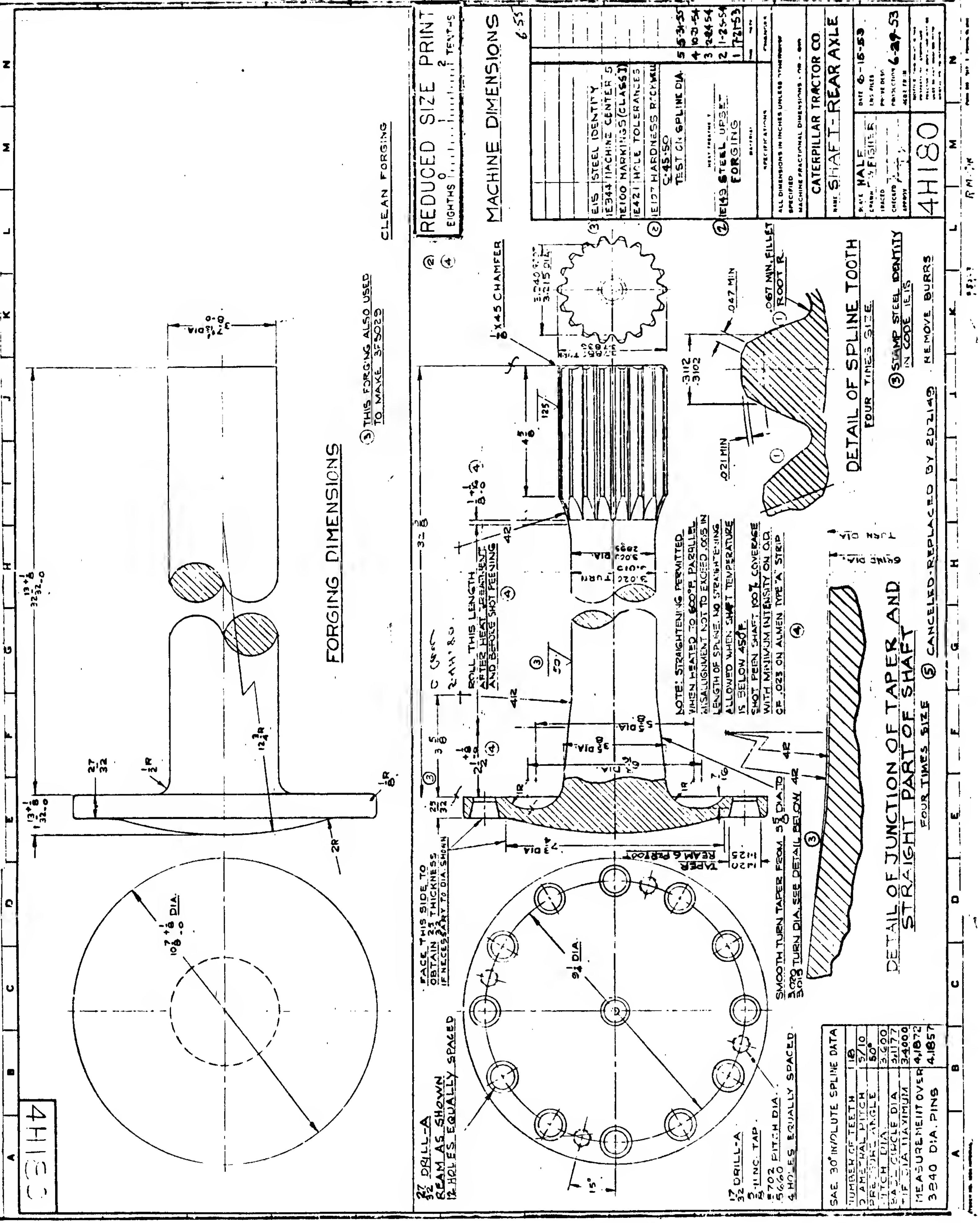
In the original design it had been contemplated that the axles would need to be strong enough to take a forward "slip" torque in fatigue but not a fully reversed "slip" torque.

At this point the hot job got even hotter. Ed knew why the shafts were failing but still the problem remained of how to improve their strength without a costly replacement of bearings and axle housings which would have to accompany the use of larger axles.

Ed decided to make a literature search and did so with the help of John Millan, a fellow engineer at Caterpillar. They found information on the testing of truck axles in bending and the effects of shot-peening on fatigue strength. (1)

A decision was made to design and build as quickly as possible a test stand that could be used to test any new axle shaft designs for their torsional fatigue strength. A photograph of the test stand which was designed and built in just a little over two months is shown in Figure 5. The machine was hydraulically operated and made up mostly from available Caterpillar hydraulic parts.

Figure 3 - Shop Drawing of DW 15 Axle Shaft



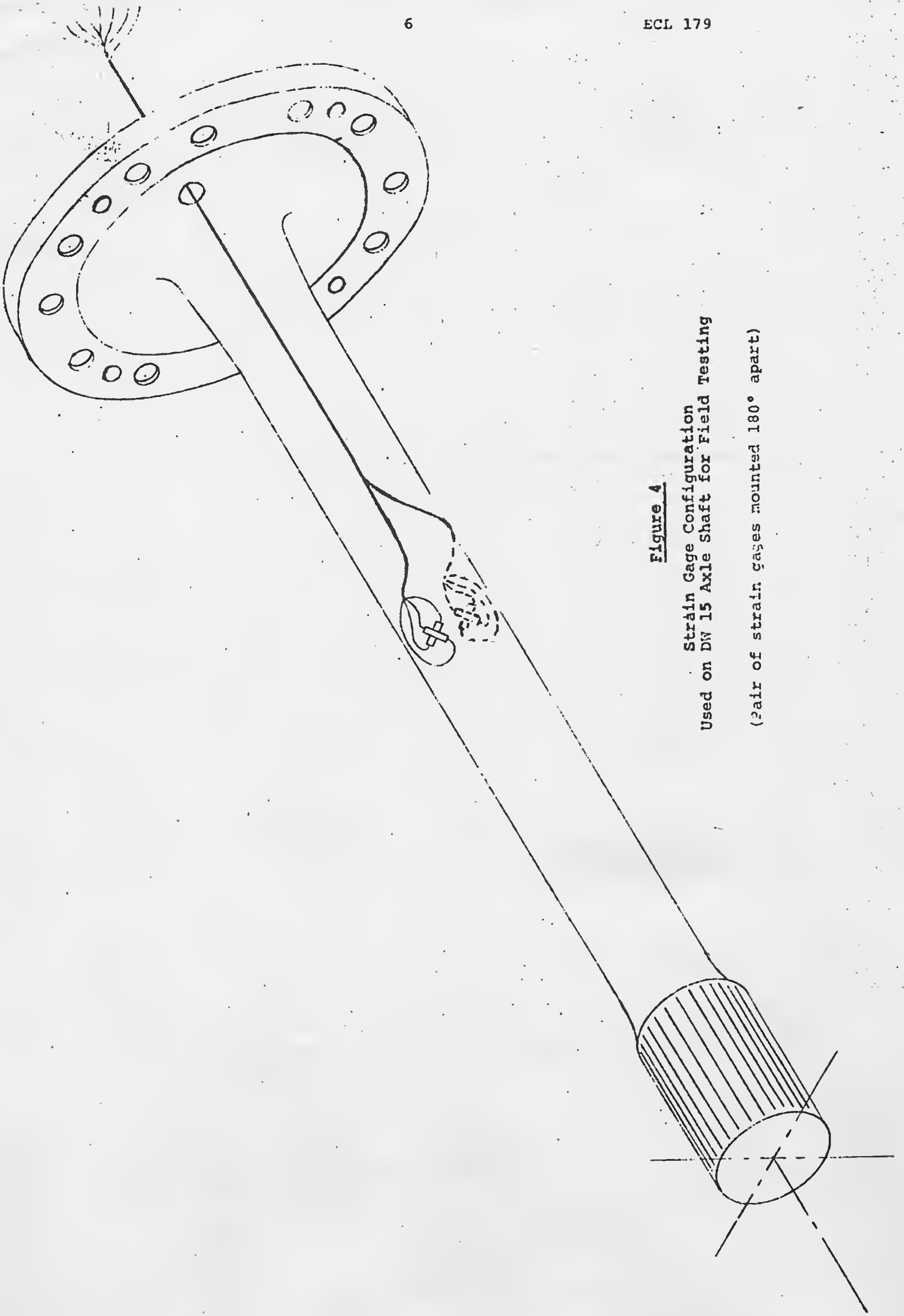


Figure 4

Strain Gage Configuration
Used on DW 15 Axle Shaft for Field Testing

(Pair of strain gages mounted 180° apart)

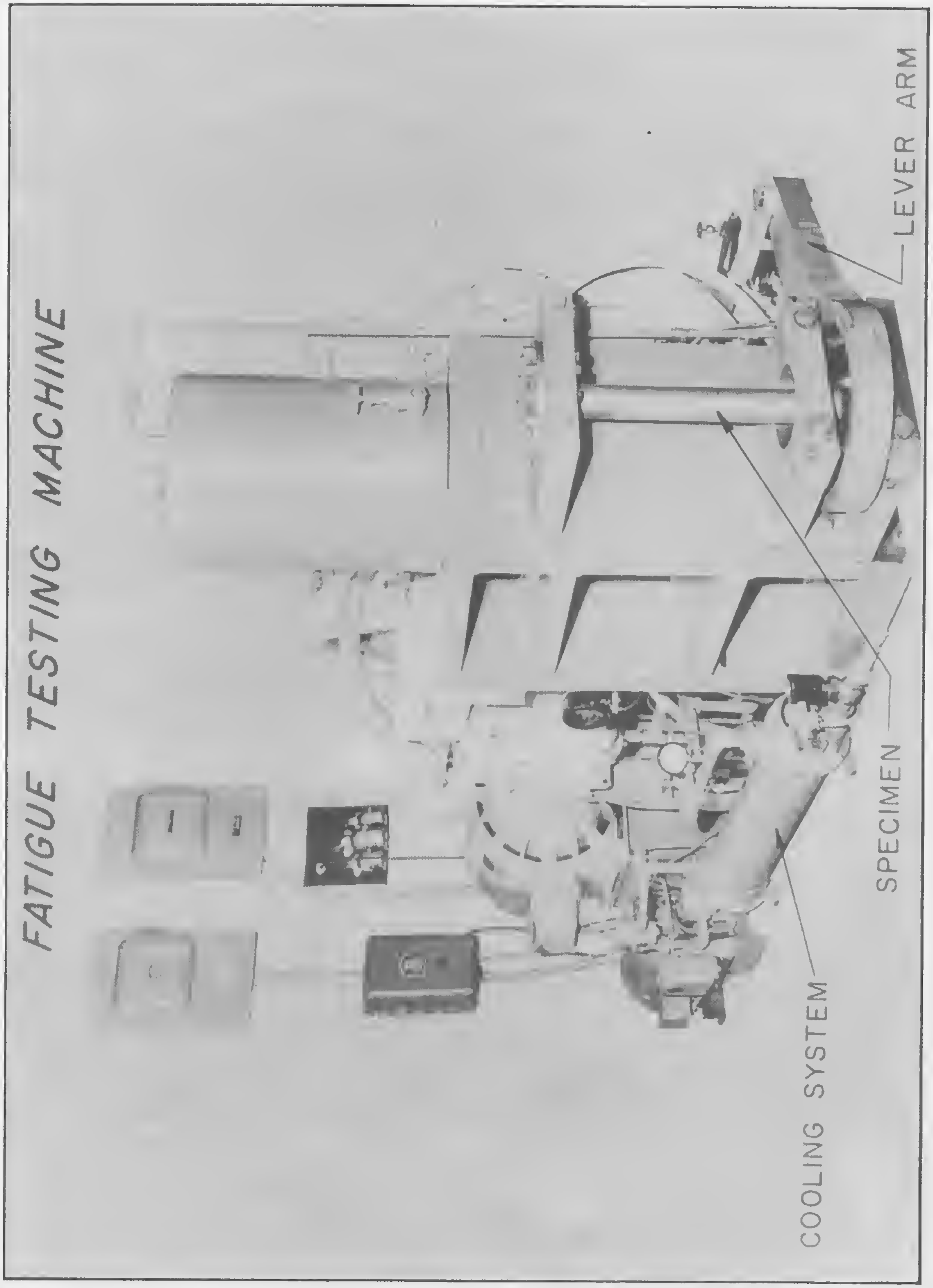


Figure 5

Torsional Fatigue Test Stand

Power to run the hydraulic pump was supplied by a 40 horsepower electric motor and an oil cooler was used to cool the working hydraulic fluid. The test machine was built in 2 months and cost in the neighborhood of $15,000.

Ed started the torsional fatigue testing of axle shafts almost immediately after the machine was completed. As a result of the fatigue data obtained with the new torsional fatigue testing machine, a decision was made to strengthen the DW 15 axle shafts by changing the material from 8645 steel to 86B45 steel. This new material was very similar to the original except for a small amount of added boron. The boron was added to increase the hardenability of the steel. The spline design was also changed to provide a full radius at the root of each spline. (See revision notes shown on the shop drawing in Figure 2.) Finally the shafts were severely shot-peened to increase their fatigue strength to the desired amount.

Because of interest in the fatigue of large shafts and the scarcity of available data, Ed published a paper about the construction and operation of the test machine and the fatigue results which he had obtained using the machine. (2)

An interesting result which Ed reported in his paper was the fatigue strength of two sets of shafts of the same material but different size. The smaller 3/4 inch diameter shafts had fatigue strengths which averaged about 140 ksi, whereas the larger 3 inch diameter shafts had fatigue strengths which averaged only 90 ksi. Another interesting result which Ed reported was the difference in life between peened and unpeened large shafts. Ten unpeened shafts had lives ranging from 17,000 cycles to 48,000 cycles of loading and 16 peened shafts of the same size and material had lives ranging from 80,000 cycles to 325,000 cycles.

The hot job was now successfully completed. The axle shafts served well, but improvements were continuously sought. The appendix shows a report concerning one such improvement effort.

REFERENCES

(1) Horger, O. J., and Lipson, C. H., "Automotive Rear Axles and Means of Improving Their Fatigue Resistance," *Symposium on Testing of Parts and Assemblies*, ASTM STP No. 72, New York, 1946, pp. 47-68.

(2) Eckert, E. J., "Torsional Fatigue Testing of Axle Shafts," *Symposium on Large Fatigue Testing Machines and Their Results*, ASTM STP No. 216, Atlantic City, 1957, pp. 21-36.

CATERPILLAR TRACTOR CO. **RESEARCH DEPARTMENT**

FILE NO. 13345 REPORTED BY E.F. RANDOLPH 5-28-57

OBSERVED BY E.F. RANDOLPH AND D.R. RICHARDSON 7-23-56 TO 5-17-57

Fatigue testing of various axle shafts

~~SHAFTS WITH NON-CIRCULAR PROFILE~~

REPORT NO. 11

EVALUATION OF 5X591 AXLES WITH HEXAGONAL SHAPED CONNECTIONS AT THE DRIVE ENDS IN DW15 TRACTORS AT THE ARIZONA PROVING GROUND

REPORT NO. 11
JOB NO. 10103-1
CHARGE NO. 93-4994
JOB SUPV. J.F. MILLAN

INTRODUCTION

The axle and final drive gear of the (RM98) DW15 tractors have always been troublesome. Measurements of axle torque showed that these parts were weak. Shot peened axles with better material and higher hardness were then made available to production. This improved the axle life to a point where the weak section is now in the spline area. A hexagonal drive connection was tried in the laboratory with marked success and four assemblies were shipped to the Arizona Proving Ground for endurance testing. These axles, after being reworked, were installed in two DW15 tractors 1A1197 (RM98) and 1A1425 (RM106) with B512R6 (5 1/8 X 6 1/2) engines set for 168 hp at 2300 rpm.

CONCLUSIONS

1. The serious wear and fretting from line contact between the 5X592 final drive gear hub and 5X591 axle must be eliminated before this assembly can be released for production.

2. The hexagonal hole in the 5X592 final drive gear hub wears too fast.

SUMMARY OF RESULTS

1. The left axle in RM98 1A1197 seized in the final drive tube just after the axle was installed (see Sheet A).

2. The right axle in RM106 1A1425 seized in the housing tube after 1677 hours as shown on Sheet B.

SUMMARY OF RESULTS (CONT'D)

3. The left 5X592 final drive gear failed through one tooth after 2187 hours.

4. The wear on hub and axle in left final drive on 1A1425 after 2187 hours is shown on Sheet C.

5. The right axle in 1A1197 was inspected after 939 hours and was in good condition except for fretting between the axle and final drive gear hub. (See Sheet A.)

RECOMMENDATIONS

1. Decrease the clearance between the 5X592 bull gear hub and the 5X591 axle to help reduce line contact for further testing.

2. Increase the hardness of the hub in 5X592 final drive gear.

<u>DISCUSSION</u>

These axles were too large to fit into the 3H9165 axle housings when received from Peoria and were reworked. This reworking consisted of grinding the axles down to a 3.788-3.798 dimension (Approx. 1/16" less than housing bore). They were also shot peened to .023" arc height on Almen A test strip.

The axles and final drive gears were then installed in RM98 (DW15) tractor 1A1197 at smr 4082 and RM106 (DW15) tractor 1A1425 at smr 2333. The left axle in 1A1197 seized in the 3H9165 housing tube just after it was assembled as the clearance between these parts was too small. The axle was cleaned up and reinstalled, but seized again after two hours. This seizure was so serious that it was necessary to remove the parts with a cutting torch as shown on Sheet A.

The right 5X591 axle and 5X592 final drive gear were inspected after 939 hours (smr 5021) and found to be in good condition. There was some fretting and wear on the axle and hub on the driving corners of the hub and axle; however, this wear was not considered serious. (See Sheet A, Photograph No. 6189.)

The right 5X591 axle and 5X992 final drive gear in 1A1425 were removed from test after 1677 hours (smr 4010). These parts were operating satisfactorily before they were removed but when removed there was evidence of seizure between the 5X591 axle and the 3H9165 tube housing at about the middle of the shaft as shown on Sheet B, (Photograph No. 6593). The hub and axle showed considerable wear and fretting on the hexagonal mating parts. This shaft was returned to Peoria.

One tooth in the left 5X592 final drive gear failed after 2187 hours in 1A1425. The mating areas of the axle and final drive gear hub were worn and fretted as shown on Sheet C. This shaft and hub were measured (see Sheet E). From these measurements, it can be seen that the wear in hub is greater than on the axle. The increase in the section across the flats at the corners on the axle indicates that there may be metal transfer from the hub to the axle. It is possible that this wear allowed enough backlash in final drive to shock load the gear train which resulted in the final drive gear tooth failure.

The 5X592 final drive gear which was used in the right final drive of this machine for 1677 hours was installed in the left side to replace the failed bull gear discussed above. The inside of this hub was measured before this installation. These measurements are shown on Sheet D.

When these parts are installed new there is only line contact between the hub and axle. This does not give sufficient bearing area and the mating surfaces start to yield. This is further aggravated by relative movement between the two parts. This situation would be relieved if the clearance between the hub and axle were reduced to a very light press fit and the hardness of the hub increased. This light press fit could result in a serious disassembly problem, especially if the serious fretting continues. It is therefore recommended that new axles be made up for test with a light press fit between the mating hub and axle. Provisions should also be made to pull the axle when disassembly is necessary. Special attention

DISCUSSION (CONT'D)

should be given to backlash between the 5X591 axle and the 5X592 final drive gear hub in future testing. This backlash could create serious gear train failures if it is not controlled.

Two axle assemblies are still on test with one assembly in each tractor. The hub and axle assembly in the right final drive of 1A1197 has operated 1904 hours to date (smr 5986). The left axle in 1A1425 has been on test for 2727 hours and the final drive gear for 2217 hours. These assemblies will be left in the tractors to accumulate more hours.

E F Randolph
E.F. Randolph

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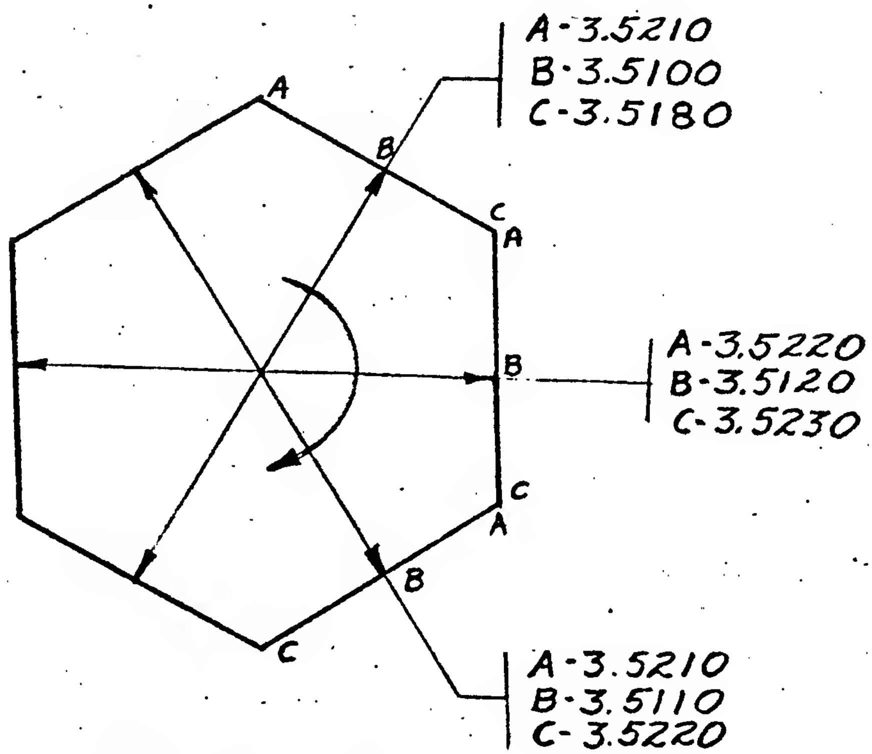
ECL 179

WEAR ON 5X592 HUB FROM 1A1425 AFTER 1677 HOURS



5X592 HUB

BY E.F. RANDOLPH

WEAR ON AXLE AND[6] FINAL DRIVE GEAR HUB AFTER 2187 HOURS

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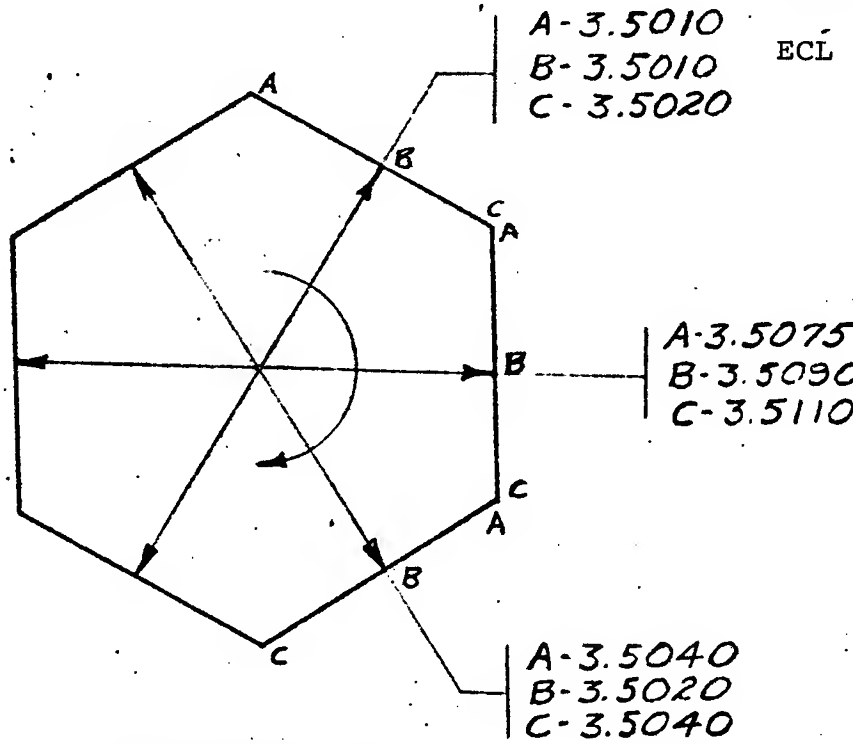
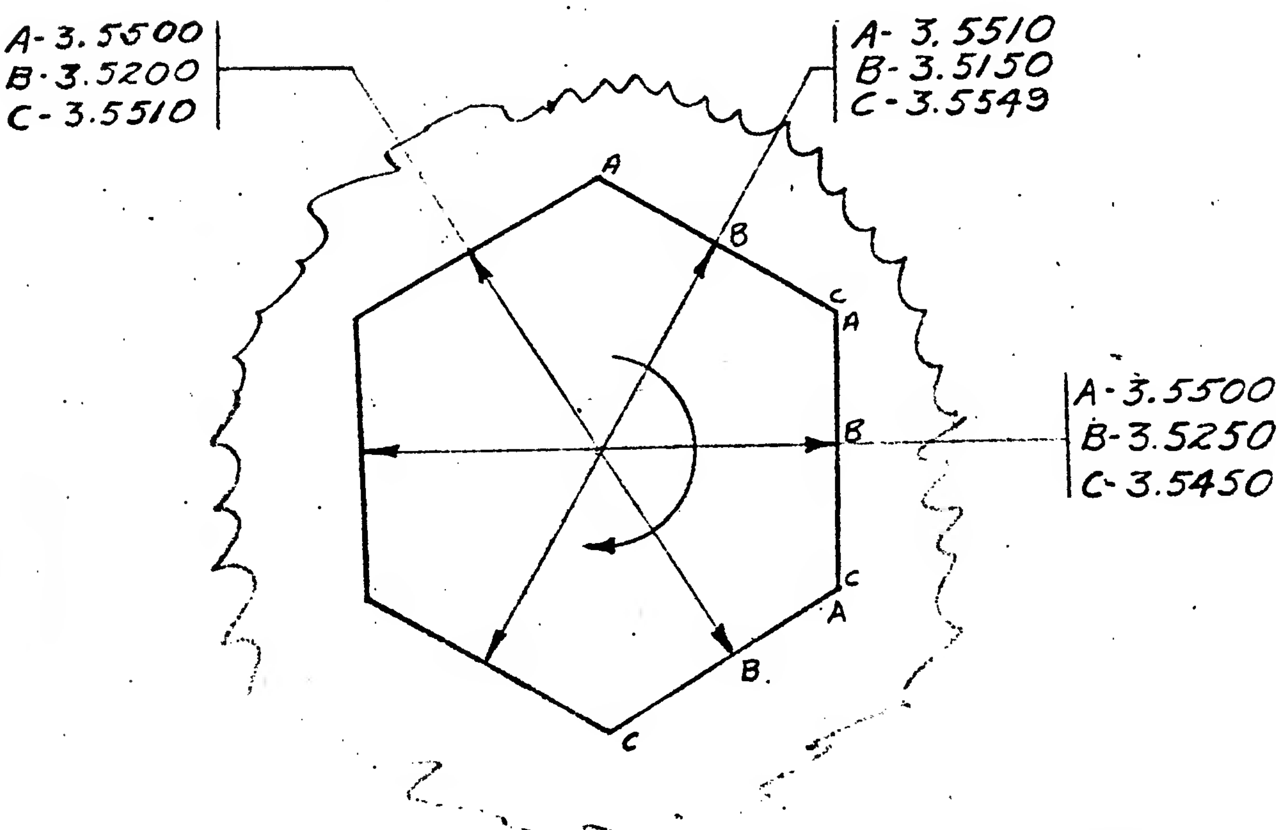


5X591 AXLE



5X592 BULL GEAR HUB

INSTRUCTOR'S NOTE

Sometimes It Gets Pushed

Possible Questions for:

Freshmen:

1. What kind of education would Ed Eckert need for a job like this?

2. What is the purpose of the 4 tapped holes in the flange of the axle shaft?

3. How much freedom do you suppose Ed had from his boss when he was given the job?

4. For the slip-torque of 50,000 ft.-lbs., what would be your estimate of the slip force (where the tire meets the ground)?

5. What would the work per load cycle be assuming that at full load the twist is $\theta = 0.2$ radians?

6. How would you improve the strength of the shaft?

Juniors:

1. Discuss why Ed Eckert used the strain gage configuration that he did on the axle shaft that he took to Baltimore?

2. What strain is measured? What stress (kind) was Ed trying to determine?

3. For 50,000 ft.-lbs. of torque what amount of shearing stress does the axle experience at its surface? What amount of shearing strain? How much twist angle?

4. What other suggestions for improvement of the axle shaft might you have?

5. If the axle shaft received 50,000 ft.-lbs of torque in one direction of rotation only and lasted for one million cycles, how many cycles do you suppose it would last if it received this much torque in both directions of rotation?

Instructor's Note (continued)

6. Why can a smaller diameter shaft of the same material withstand a higher stress in fatigue than a larger diameter shaft?

7. Discuss the dynamics of the torque reversal in the axle shafts when the DW 15 gets pushed. How can the wheels slip and the reverse torque get so high?

8. Where does the heat come from which is dissipated by the oil cooler on the test stand?

Graduates:

1. Estimate the cost of an axle shaft failure for:

   A. The customer (tractor owner)
   B. The Caterpillar Tractor Company

2. What might be some other ways to improve the fatigue strength of the DW 15 axle shafts?

3. What do you suppose would make the peened shafts last longer? Explain the difference in observed lives.

4. For what fatigue life should the designer aim for the DW 15 axle shafts under 50,000 ft.-lbs. of fully reversed fatigue loading?

5. Discuss the differences, advantages, and disadvantages between field tests, proving ground tests, and lab tests.